

USING 3D MULTI-BODY SIMULATION TO EVALUATE FUTURE TRUCK TECHNOLOGIES

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ABSTRACT

This document presents the results of computer-based, vehicle dynamics performance assessments of Future Truck concepts with such features as a variable height, hydraulic, trailing arm suspension, skid steering, and in-hub electric drive motors. Fully three-dimensional Future Truck models were created using a commercially available modeling and simulation methodology and limited validation studies were performed by comparing model predictions with baseline, validated model predictions from another vehicle in the same size and class as the Future Truck concept vehicles. The models were considered accurate enough to predict various aspects of ride quality and stability performance, critical to US Army Objective Force mission needs. One-to-one comparisons of the Future Truck concepts and a standard, solid-axle, Heavy Tactical Vehicle (HTV) operating in various terrain and obstacle negotiation conditions were performed.

INTRODUCTION

The US Army vision, announced in October of 1999, encompasses people, readiness, and transformation. The goal of the Army vision is to transition the entire Army into a force that is strategically responsive and dominant

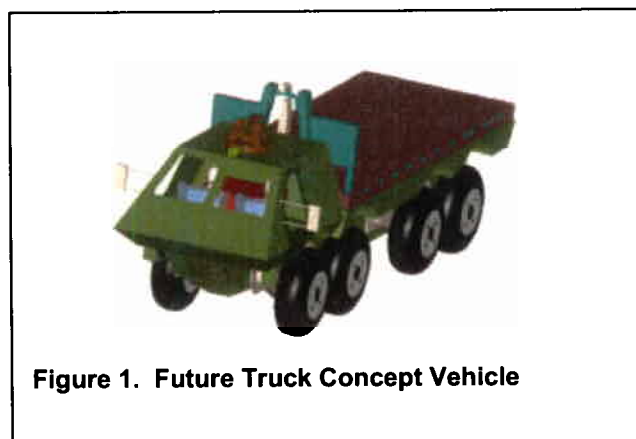


Figure 1. Future Truck Concept Vehicle

at every point of the spectrum of operations. The transformation component will be accomplished in three ways: the current Force, the Interim Force, and the Objective Force. The objective force is not platform driven, but rather the focus is on achieving capabilities that will operate as a "system of systems." As part of the Objective Force, the US Army plans to begin the fielding of selected Future Combat System (FCS) in FY08 [1]. As part of the FCS program, the Tactical Wheeled Vehicle Modernization [2] encompasses all US Army tactical wheeled vehicles. The US Army's Tank Automotive Research, Development and Engineering Center (TARDEC) has the task of identifying technologies and capabilities that will make the FCS challenges a reality. TARDEC's National Automotive Center (NAC) is using modeling and simulation to demonstrate the feasibility and operational potential of advanced commercial and military technologies with application to new and existing tactical vehicles and to describe potential future vehicle capabilities.

The objective of this paper will be to describe how M&S is being applied to answer a wide variety of design and performance evaluation questions. It depicts a series of simulation-based engineering projects that build on the Army's simulation investments as a tool to investigate and answer real-world vehicle design, acquisition, and

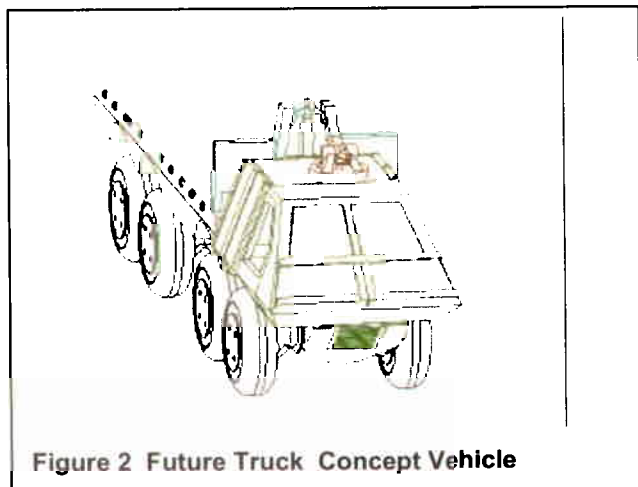


Figure 2 Future Truck Concept Vehicle

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engineering support questions. Due to much increased High Performance Computing (HPC) computational speeds, memory, and asset availability, entire spectrums of operational mission scenarios are investigated and simulations conducted over a wide range of vehicle speeds and operating conditions. Recent major upgrades in HPC facilities now allow the highly detailed, computationally intensive models to be run in a fraction of the time, and, more importantly, many more 'what if' studies are being performed. Using HPC-based vehicle performance modeling & simulation in support of acquisition allows the Army to evaluate the performance of numerous proposed vehicle system configurations analytically, thereby saving time and costs associated with building and testing actual prototypes. The NAC's M&S efforts using HPC is constantly striving to make the Army a smarter and more cost-effective buyer of equipment, and more importantly, significantly reducing the associated risks that are inherent in procuring newly designed, untested equipment.

The NAC serves as the Army's agent for advancing the development of dual-use automotive technologies by industry, academia and the military services. By cultivating relationships and forming cost-shared partnerships, the NAC accelerates the exchange and implementation of automotive technologies creating developmental savings that are shared by all participants. The U.S. Military requires flexible, effective and efficient multi-mission forces capable of projecting overwhelming military power worldwide. To satisfy this requirement, the FCS program was developed to provide enhancements in land force lethality, protection, mobility, deploy-ability, sustainability, and command and control capabilities. The goal of the FCS program is to design an ensemble that strikes an optimum balance between critical performance factors, including ground platform strategic, operational and tactical mobility; lethality; survivability; and sustainability. This "system of systems" design will be accomplished by using modeling and simulation and experimentation to evaluate competitive concepts as defined in the FCS simulation support plan [3]. The FCS will be capable of adjusting to a changing set of missions, ranging from war-fighting to peacekeeping as the deployment unfolds.

What sort of tactical wheeled vehicles will be needed to support the Army in the next decade? To provide accurate answers to that question, the NAC created the Future Truck concept (see figures 1 & 2) to investigate critical technologies that will be required to achieve stated FCS goals and objectives. Some of the technologies to be explored in this analysis to enhance stability, handling and mobility include; active-variable height hydro-pneumatic suspensions, advanced hybrid electric propulsion systems, electronic steering, and central tire inflation.

FUTURE TRUCK CONCEPT REQUIREMENTS

NAC engineers are using three-dimensional illustrations, models and simulations to help optimize the survivability, mobility, transportability and supportability of the Army's future tactical trucks. A combination of virtual prototypes and critical hardware demonstrations will leverage the Army's technology programs, and those from the commercial sector to revolutionize future logistics support. The US Army Objective Force mission requirements will include a heavy tactical vehicle capable of transporting in excess of 10 ton cargos on-, and off-road.

The Army's Future Trucks are expected to include a load handling system which will be capable of transporting 4 and 8 foot ISO containers. In order to keep up with the speed of, and provide logistic support for the Future Combat System, the operating requirements for this vehicle are expected to be much more severe than equivalent payload class vehicles in today's fleet. The lighter overall system weights and increased performance make the development of a survivable and reliable Truck an engineering challenge. The concept technologies being investigated as part of this study include independent, active and passive, variable height, hydro-pneumatic trailing arm suspensions, skid steering, hybrid-electric power, and in-hub permanent magnet electric drive motors for vehicle system propulsion that runs silent, and can provide braking. Also included in the analysis was an advanced, lightweight, 450 horsepower diesel engine to keep the batteries charged and provide power to the system as required during operations, as well as large 16xR20 radial tires with run flat inserts and central tire inflation (CTIS) to maximize off road mobility and payload carrying capacity. This analysis utilized simulation tools and methodologies that lead to design options that are able to meet Army goals. The NAC objectives are to develop and mature the processes necessary to evaluate new technologies and optimize the decision-making processes.

APPROACH

NAC engineers applied a commercially available "Dynamic Analysis and Design System" or DADS [4] modeling and simulation methodology to simulate the performance of the Future Truck concept technologies. These models are suitable for obtaining a better understanding of the vehicles' performance characteristics and for assessing future technology upgrades that could lead to better performance. Future Truck technology demonstrator models include accurate representation of all suspension components to provide predictions of relative displacement between subsystem components. Nonlinear models of suspension stiffness and damping, jounce and rebound stops and steering stops were incorporated to provide accurate interaction force predictions. Individual rotating wheels with nonlinear tire/terrain interaction models that allow the wheels to leave the surface were included in the model to allow large vehicle displacements, including rollover. Rolling tire models that generate fore-aft tractive and side-to-side forces between tire and terrain were included to insure

representative mobility predictions. Second order steering and speed control algorithms were used to keep the vehicle model on course and to maintain desired speed.

Conceptual Truck models were loaded to 10 tons and executed over a number of artificial obstacles defined by NAC engineers. The purpose of these short duration, transient maneuvers was to provide repeatable disturbance inputs to the model with well controlled initial conditions. The models were also executed over a number of straight-line symmetric and non-symmetric variants of the Perryman 2 and 3, and Churchville B courses located at the US Army's Aberdeen Test Center. The simulations were conducted at various speeds to induce different levels of response, and to investigate the upper limits of safe operational performance.

A model of an existing Army heavy tactical vehicle was developed and loaded to 10 tons and executed over the same obstacles, maneuvers and courses as the Truck concepts, to identify and baseline safe speeds for operation through the various operational scenarios so one-to-one comparisons could be made between the vehicle systems in order to quantify any performance gains attributable to the technology upgrades and or insertions. Computer-generated animations of each simulation were made and recorded for review in real time.

This paper first describes how a representative Concept Truck model was defined and developed. A general overview is provided of the topology, parameters and performance characteristics of each major subsystem. Then we provide an assessment of how each subsystem influences the vehicles' operating performance envelope. Details of the existing Army HTV model are given in [5] and are not presented here. A general overview of the modeling strategy and a description of each major vehicle subsystem model are given to provide a better understanding of the composite vehicle model operation, its interaction with the obstacle and terrain models, and the simulation results. Data and descriptions of each obstacle, maneuver, course profile, as well as a matrix of the maneuvers and course negotiations are presented, and simulation results are summarized.

VEHICLE DATA

Figure 3 shows a computer generated graphical representation of a Future Truck concept model which contains the eight by eight Truck suspension and chassis. The primary purpose of the Truck is to transport heavy loads up to (possibly in excess of 10 tons) cross country on rough terrains at relatively high speeds, and to maintain mobility, ride quality and dynamic stability while

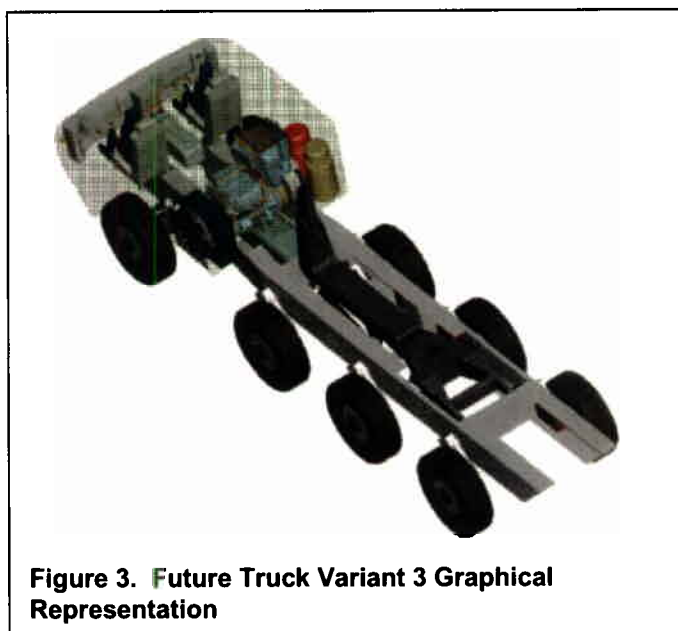


Figure 3. Future Truck Variant 3 Graphical Representation

doing so. Existing commercial and Government vehicles have not been designed to meet the rough terrain speeds required by the Future Combat System. The US Army funded the NAC to develop and apply models of the Future Truck Concepts, integrated with high potential new technologies, to assess their influences on overall vehicle mobility, ride quality, dynamic stability, and overall performance. TARDEC's advanced vehicle concepts group provided much of the Concept's vehicle parameters. Representative tire and run-flat data were acquired from the manufacturers. The rest were calculated, estimated, or developed by the NAC team working the project. Active springing and damping control algorithms and data, jounce and rebound stops, roll stiffness and roll center characteristics were all calculated based on payload to weight ratios and mission requirements. Lastly, algorithms were developed to electronically control steering and wheel hub drive torques based on existing data for electric motors currently available.

FUTURE TRUCK CONCEPT MODEL STRUCTURE

As depicted in figure 3, the evaluated concept's chassis contains a rigid ladder frame along the full length of the vehicle that each of the trailing arm units attach to, along with the corresponding wheel hub and electric motor assemblies. Each trailing arm rotates up and down to allow the necessary suspension displacement with a jounce and rebound stop installed at each wheel station to prevent excessive suspension travel. All wheel stations have hydro-pneumatic suspensions to provide the necessary support and damping. Skid steering was modeled by controlling the electric drive motors in each wheel to control the "left side" versus "right side" wheel speeds to develop vehicle yaw torque to steer the truck. This steering arrangement allows the wheels to be

electronically controlled and was performed by a 2nd order non-linear algorithm.

The vehicle chassis and payload are each represented by rigid bodies. Eight rigid bodies representing the eight trailing road arms (with hydro-pneumatic suspension unit masses included) are connected to the chassis body at the physical pivot point locations in the vehicle through transversely orientated revolute joints. Eight wheel hubs are connected to the trailing road arms similarly through transverse revolute joints.

SUBSYSTEM MODELS

The concept vehicle model contains a number of hard and soft mechanical stops that must be adequately represented to insure proper inter-component displacement predictions. A nonlinear translational spring between the chassis and each trailing road arm was used to represent the corresponding jounce stops. These springs were placed in line with physical stops that could be mounted on the chassis, and the stiffness of these jounce stops were set to known values for thick rubber jounce stops used on vehicles of this size and weight class.

ACTIVE SUSPENSION CONTROL ALGORITHM

The active suspension control algorithm was developed by NAC engineers working this project, and it was developed based on the premise (Patent 5,999,868 constant force suspension) of University of Texas-Austin work on active suspension in that we introduced what we termed as "bias" springs to support the vehicle's weight. These "bias springs" were somewhat artificial in that they had a zero slope force deflection curve (hence the term "bias"). The forces generated by the active suspension's were those computed forces required to keep the relative chassis vertical displacement zero. To keep this scheme from failing while ascending or descending hills, averaging the suspension station displacements was employed to identify an additional spring force which was then added to the hydro-pneumatic force which effectively corrected for the hilly trends of the terrains.

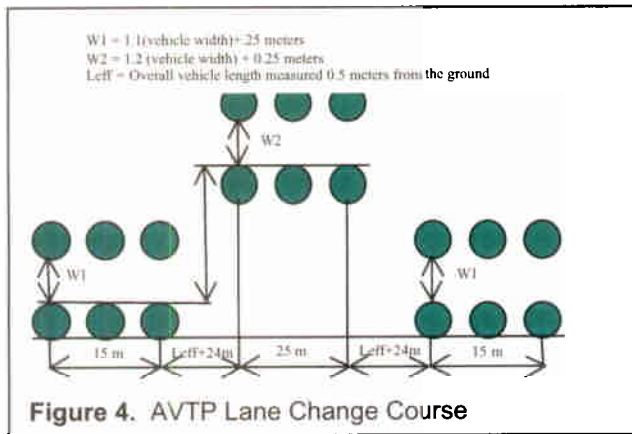
The tires were modeled using the DADS full tire model and the tires' restoring forces were modeled by nonlinear force deflection curves that allow for the interaction forces to go to zero when they lift off the ground. The vertical force developed between the Truck tires and a non-deforming ground surface as a function of vertical tire deflection was taken from tests performed on similar tires. Additional data causing the curve to increase nonlinearly to emulate hardening effects due to bottoming out on the rims was included. The vertical stiffness rates corresponded to cross-country tire inflation rates and also included stiffening (due to CTIS) to develop on-road tire forces for handling and dynamic performance tests. The tire model computes the relative slip velocity between the bottom of the tire and ground and breaks this relative slip

velocity into the lateral slip component and longitudinal slip components. The longitudinal slip velocity is used to compute a fore-aft slip between zero and 100 percent. This computed slip was used to compute the coefficient of friction based on the tire's measured longitudinal slip characteristics. The coefficient of friction is multiplied by the tire normal force to obtain the longitudinal frictional force. The lateral slip angle was computed by subtracting the wheel yaw angle from the absolute direction of travel of the center of the tire in the ground plane. This slip angle, the tire pressure, and the tire normal force are plugged into a tire carpet plot table lookup routine that outputs the corresponding lateral force. The vertical, longitudinal, and lateral forces were applied to the wheel bodies to represent the forces generated by the tires.

The speed of the vehicle is determined by projecting its velocity vector along the chassis fore-aft centerline. This result is compared to the desired speed and a corrective torque is generated. This torque is applied to each wheel hub to propel (or brake) the corresponding wheel, which effectively controls the vehicle motion. A simple steering algorithm monitors the vehicle's centerline alignment with, and its deviation away from a designated trajectory. These two error signals are converted to a velocity dependent steering torque that is applied at each wheel to steer the vehicle. The gain in this controller model was made inversely proportional to vehicle speed to reduce steering sensitivity at higher speeds for better steering stability.

TEST SCENARIOS

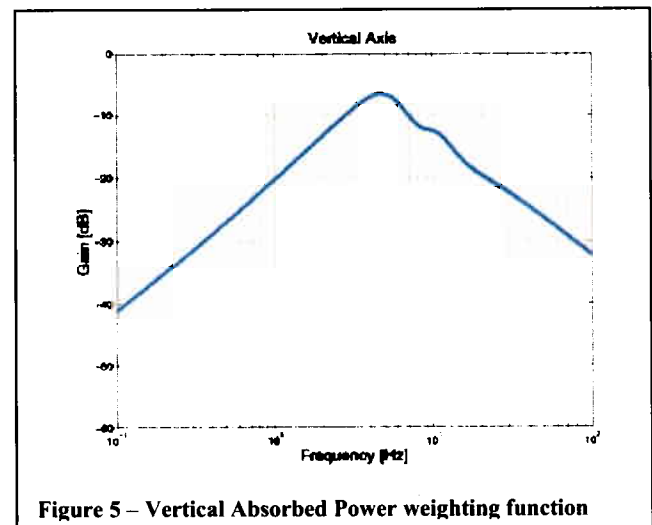
The Future Truck concept models and the HTV model were loaded to 10 tons and were executed over a number of artificial pothole and bump obstacles. Each obstacle was designed so only the left side tires encountered the obstacle in order to induce significant nonsymmetrical transient responses. Lane change and obstacle avoidance maneuver simulations were also conducted to evaluate the vehicles' lateral stability and handling characteristics. The purpose of these short duration, transient maneuvers were to provide repeatable disturbance inputs to the model with well controlled initial conditions. The model was also executed over straight-line symmetric and asymmetric variants of the Perryman 2, Perryman 3, and Churchville B courses measured at the US Army Aberdeen Test Center, Aberdeen Maryland. In addition to these measured terrains, artificial courses were generated by shifting the left track 9 feet ahead of the right track to simulate non-symmetrical terrain inputs to the vehicles. It is important to note that the actual terrains measured in the field contain turns that were not included in our analyses. The results should be used primarily for making comparisons between the two vehicles as they would perform on these artificial course segments. The simulations were conducted at various speeds (dependant on the surface roughness of the terrain) to induce different levels of response, and to investigate the upper limits of safe operational performance.



In addition to the aforementioned cross country terrains, artificially constructed ramped bumps, half-round bumps, potholes, and a double lane change maneuvers were programmed and negotiated by the modeled systems. These simulations were performed on various ramp/half-round heights and pothole depths. The maneuvers were designed so only the left side wheels of the trucks encountered the ramped bumps and potholes, while both sides of the vehicles wheels hit the half round bumps to evaluate shock performance. The gate locations for the Allied Vehicle Test Procedure (AVTP) double lane change maneuver were programmed and executed in order to evaluate the dynamic lateral stability and handling of the vehicle systems. The double lane change maneuver requires the vehicles to execute a lateral transition to the left and then back to the original lane as shown in figure 4.

VARIABLES RECORDED AND SIMULATIONS CONDUCTED

In order to determine the Concept Truck's stability characteristics, we included several cross-country courses in the test matrix. These cross country courses varied in roughness (what are they) and also in the amount of roll they induced. In addition to these cross-country courses, the test matrix included double lane change maneuvers. These maneuvers were instrumental in determining a vehicle's Lateral Acceleration Threshold (LAT). The LAT is the highest lateral acceleration a vehicle can withstand without exhibiting tire lift-off. Because we were performing comparisons between vehicles, rather than tabulating a list of lateral accelerations, we ran the lane change simulations at speeds high enough to cause one vehicle to roll over. In both cases, the HTV rolled over at 64 km/h and the Concept Truck remained upright. The test matrix also included several (how many) pothole and bump obstacle courses. These courses excite the vehicle system with a single discrete event, which enables us to easily compare the effects of suspension jounce clearance by measuring the percent of time on jounce stops. Though not as



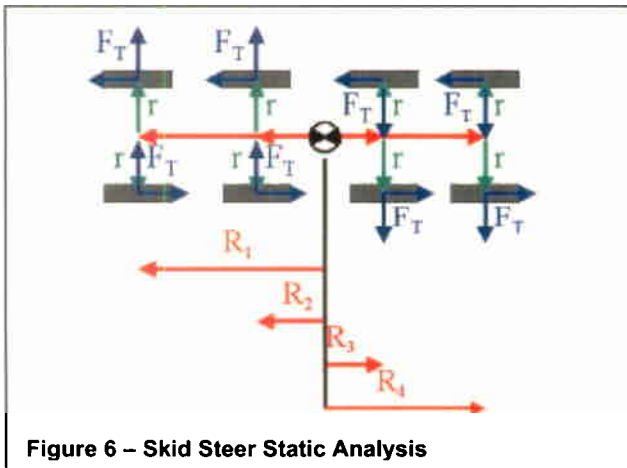
pronounced as the cross-country courses, the courses do impart significant roll and pitch motion to the vehicle system, thereby allowing us to compare vehicle roll and pitch compliance. Table 1 gives a summary of the different simulations performed.

RIDE QUALITY EVALUATION

In the late 1960's, Army researchers Lee and Pradko developed the ride quality performance criterion known as absorbed power [6][7][8][9][10], the time average of frequency weighted acceleration squared, defined in Equation 1 as:

$$P_{absorbed} = \frac{1}{T} \int a_w(t)^2 dt \quad (1)$$

Where $a_w(t)$ is the recorded acceleration at the driver's station, and weighted by the filter whose characteristics are shown in Figure 5. Army user jury tests have identified experimentally that the average value for sustained vehicle operation on average is 6 watts absorbed power. These jury studies also indicate that 15-20 watt levels are sustainable for short periods of time. This work was used in part in the development of International Standards Organization (ISO) 2631 - Evaluation of human exposure to whole-body vibration for periodic, random and transient vibration. Both filters show high sensitivity to the natural frequency of the thorax-abdomen in the 3-4 Hertz range. Smith et al (1978) provides a comparison between the two with similar results.

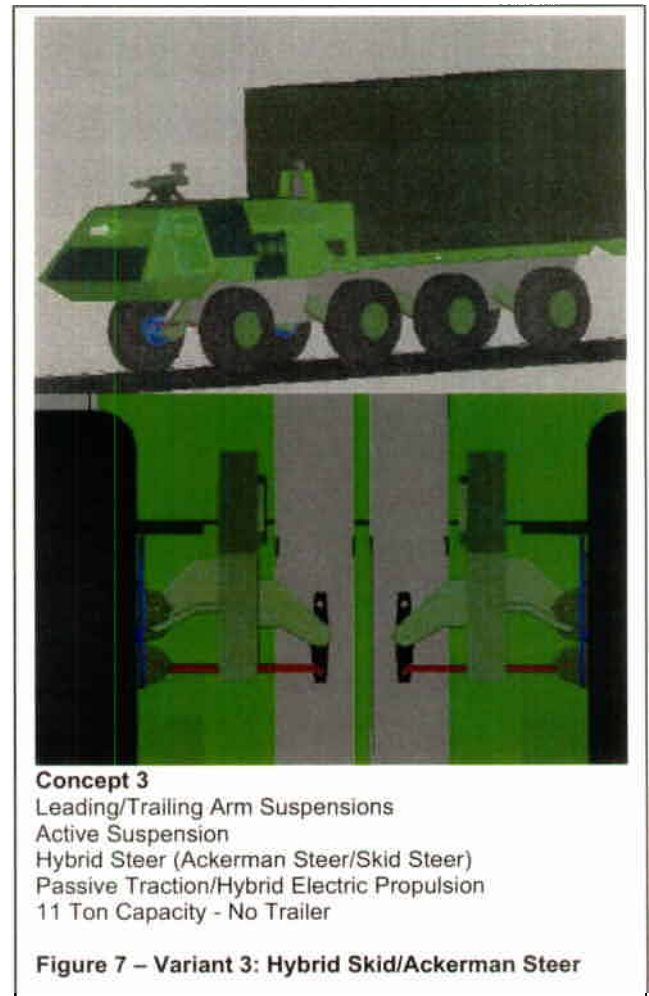


We have adopted Lee and Pradko's absorbed power measurement techniques to predict ride quality performance of concept vehicle systems as well as systems undergoing upgrades or other configuration changes. Typically, curves of 6 Watt limiting speeds over various terrains are generated and compared to identify ride performance trends or improvements. Improvements in ride quality performance result in raising the 6 Watt limiting speeds over terrains. These criteria have been applied to tune vehicle design parameters such as springing and damping rates in efforts to optimize system ride quality and dynamic stability performance of virtual prototypes. Parametrically tuning suspension parameters gives vehicle designers the opportunity to evaluate system performance in simulation before making investments in hardware prototypes.

Course Description (m=meter)	Speeds (km/h)
Churchville B - 152 m	16, 32, 48
Perryman 2 - 152 m	16, 32, 48
Perryman 2-152 m	16, 32, 48
Perryman 3-152 m	16, 32
Perryman 3-152 m	16, 32, 48
lane change-3.3x27.4 m	32, 48, 64
pothole- 2286 m deep	32, 25
pothole- 3048 m deep	16, 32, 40, 48
bump- 4572 m high	24, 32
bump- 6096 m high	16

Table 1. Simulations

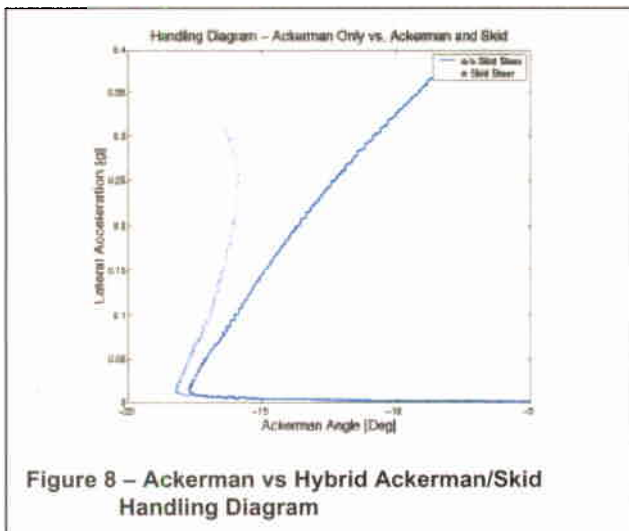
We applied this criteria to all three Variants modeled to date over a 3 inch RMS roughness terrain, and compared these 6 Watt speeds to that of the HTV over the same terrains. The Variant 2 Truck model with passive independent double wishbone suspension had a 6 Watt speed of 24 mph over the 3 inch RMS terrain, while the HTV had a 6 Watt speed of 16 MPH. The addition of active control to the independent suspension resulted in 6 Watt speeds of about 32 MPH. Similar 6 Watt speeds were obtained for all the actively controlled independent suspensions modeled, whether double wishbone or road arm type. Greater 6 Watt speeds can be expected for a



better optimized, and or adaptive, actively controlled suspension.

STEERING AND HANDLING

Concept designers had to consider several trades when arriving at the design of the Concept variants; for example, when considering Hybrid Electric propulsion (series hybrid and parallel hybrid), they had to take into account battery pack volume, weight and location. The power required to propel a vehicle of this size (200-300 kilowatts), with today's battery technology requires considerable volume for battery storage. Locating the battery packs as low as possible to reduce the overall system center of gravity height is of obvious benefit. To maximize the available volume available for payload and subsystem placement, vehicle designers eliminated steering linkages on Variant 1, going instead with skid steer and trailing road arms, leaving the entire volume between the road arms open for cargo and vehicle subsystems (such as batteries). The best way to demonstrate the difficulty skid steering a vehicle of this length versus width presents is to look at the forces and torques required to skid steer as depicted in Figure 6.



The tire friction force (FT) is a function of each tire's normal force and the coefficient of friction μ for the tire/ground interface. And for this analysis we can consider the coefficient of friction as being nearly equivalent in both the lateral and longitudinal directions.

$$FT \sim \mu \cdot (\text{Normal Force})$$

The maximum turning torque (TT) generated by the differential wheel torques being applied to the left and right sides respectively is a function of the lateral distance r from the vehicle centerline to each wheel, and the tire friction force FT.

$$TT \sim 8 \cdot (r \times FT)$$

The Torque Resisting Turn (TRT) working opposite to the turning torque is a function of the fore aft distance from the vehicle centerline to each wheel R_n , and the tire friction force FT.

$$TRT \sim (R_1 \times FT) + (R_2 \times FT) + \dots + (R_8 \times FT)$$

Since on average the distances R_n are greater than r , then TRT is greater than TT, therefore skid steer doesn't work for a vehicle that is much longer than it is wide.

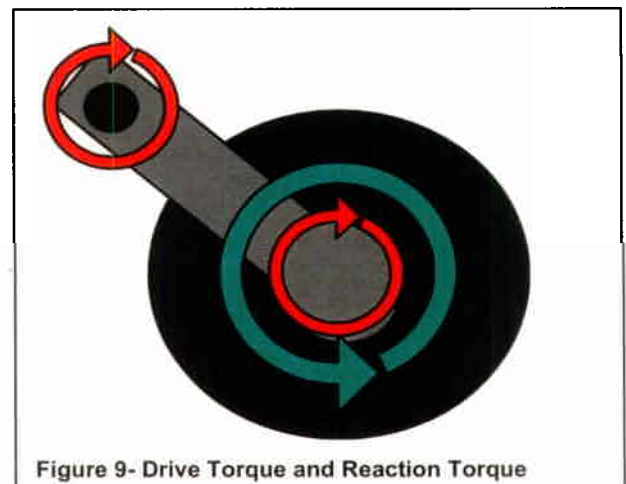
HYBRID SKID/ACKERMAN STEER

This raises the question of whether skid steer may be able to provide some benefit to an Ackerman steer arrangement. We modeled Variant 3 with trailing road arms at the rear 6 suspension stations, and with leading road arms with a modified Ackerman steer arrangement at the front suspension stations (see figure 7), and performed a yaw stability analysis on this vehicle with and without skid steer to see if there was any benefit to be

gained by adding skid steering to an Ackerman steered vehicle.

Oversteer is a condition where the rear tires exhibit greater lateral slip than the front tires while undergoing those forces due to the lateral accelerations of turning. A vehicle requiring a reduced steering correction to maintain its course as it undergoes this increasing lateral acceleration (the rear axle is slipping more than the front axles) is said to be an oversteering system. The reason oversteer is undesirable is because when the rear tires are slipping more than the front tires, if the driver does not make a steering correction, the vehicle would continue to wind an inward spiral, ever decreasing the instantaneous turning radius, until the vehicle eventually rolled over.

There are two standard methods which are typically used to gauge vehicle yaw stability, both requiring instrumenting the vehicle (or vehicle model in our case) to record the vehicle speed and Ackerman steering angle. The Ackerman angle is plotted against vehicle lateral acceleration as the vehicle undergoes increasing lateral acceleration values. If the Ackerman angle increases as the lateral acceleration increases, the vehicle is understeering. If the Ackerman angle decreases as the lateral acceleration increases, the vehicle is oversteering. The lateral acceleration is increased by traversing a constant radius turn at slowly increasing speeds, or by controlling the vehicle to negotiate an inward spiral maneuver at a constant speed. Plotting the Ackerman angle against the system's local lateral acceleration



results in what are known as "Handling Diagrams". From the slopes of these diagrams, one can identify whether a vehicle has under-steer, neutral steer or over-steer characteristic tendencies. The "virtual" test we employed consisted of controlling the vehicle's steering to follow a circular path whose radius was 30.5 meters, and slowly increasing the vehicle speed until the lateral acceleration exerted on the vehicle caused a tire to lift off the ground. The outputs from these simulations are used to generate handling diagrams.

All of these trucks have one thing in common in that they all have system centers of gravity shifted rearward of the

neutral steer static margin, therefore the lateral acceleration developed while turning acts at the rearwardly shifted center of gravity, which acts to make these systems inherently oversteering. This analysis was performed to identify whether the addition of skid steer would improve could bring these systems from an oversteer condition closer to neutral steer or to even understeer characteristic.

Some work has been done which identified the gains to handling performance resulting from increasing the lateral forces developed by the rear tires [11]. This work identified reductions in oversteer tendencies resulting from increased tire/ground contact patch area and when using dual tire configuration on rear axles versus single tires. Generating a yaw moment directed oppositely to the driver's steering input, in theory could require an increase in the Ackerman angle necessary to complete a steady state turn at increasing levels of lateral acceleration, thereby reducing oversteer. On Figure 8 we see that both the Ackerman and the Hybrid Ackerman/Skid steer vehicle both have positive slopes indicating oversteer, but we see that the Hybrid steered vehicle's slope is only slightly oversteer at low lateral accelerations, and switches to the desired understeer characteristic at higher lateral accelerations.

HUB MOTOR, ROAD ARM AND ACTIVE SUSPENSION CONTROL

If the hub motor reaction torques (drive and brake) are allowed to pass through the road arms, then these reaction torques must be countered by the active suspension control (Figure 9). In simulation it is a considerably simpler task than in the physical world because the drive torque is known at all times. The controller can be programmed to simply add it to the active control to counter the reaction. In the physical world, the active control as we have implemented it would require much greater control torques to compensate for the drive torque, this would require more power to the controller. An alternative is to mount rotating struts from the hub motor to the vehicle chassis that are positioned so as to traverse the same arc as the road arm, effectively canceling the drive torque through the hull instead of the road arm.

CONCLUSION

The most obvious result of this analysis was that the Concept Trucks completed the entire test matrix without rolling over, whereas the HTV rolled over on several cross-country courses and lane change maneuvers. Assessments of the animations and data indicate that the Concepts have superior performance over the HTV in the US Army mission environments. The main reason the Concept truck performance was better than the current vehicle is due to the active suspension. But in addition to this is that the HTV payload was proportionally higher than the load on the concept. This may have contributed to

give the Concepts a lower center of mass, and the increased inertia gave it lower roll, pitch and jounce natural frequencies than the HTV. These, plus the optimized damping ratios, may have combined to improve overall system performance. These simulations showed that both vehicles performed well in the harsh operating environments. Any vehicle designed similar to the Concepts and operating in a similar payload range should exhibit excellent dynamic performance characteristics for the types of off-road operations expected in the field.

The major performance areas which are impacted most by high resolution vehicle simulations are mobility, stability, reliability and safety. Ground vehicles operate in very harsh environments and are expected to perform their intended missions. Vehicle developers are responsible for setting realistic performance specifications and ensuring that they are met. Modeling and Simulation are making the Army a smarter and more cost-effective buyer and tester of vehicles and equipment, and more importantly, significantly reducing the associated risks to personnel and property that are inherent in a war-fighting environment. In addition, M&S can alleviate and/or avoid the endless build-test-break-fix cycles, common in many vehicle acquisition and testing programs of the past, thereby reducing costs and shortening milestone schedules. Although it is difficult to quantify overall life cycle impacts resulting from doing it right the first time, M&S has proven itself to be an excellent tool for decision makers, in response to limited time and budget constraints, as well as, aggressive procurement schedules.

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